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Original Article

OPTIMIZATION OF MEDIUM BUS FRAMEWORK STRUCTURE WITH DYNAMIC PERFORMANCE LIMITS

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ABSTRACT

In several decades, urban areas around the world are increasingly dominated by vehicles, one of which is buses. The safety and comfort factors of passengers are the main considerations for a bus driver in carrying out their activities on the highway. Researchers used an ISO-2631 curve to evaluate dynamic characteristics and predict passenger and driver fatigue limits to avoid accidents. The conceptual design stage of the bus structure is a way to meet the requirements in terms of bus performance, namely passenger comfort. Therefore, many researchers have carried out structural optimization to get the lightest vehicle weight possible without reducing vehicle performance. This study aims to obtain an optimal bus frame structure design by considering passenger comfort as a limitation of sizing optimization on the bus frame structure. The steps taken are to obtain the RMS value from the PSD Acceleration obtained from the random response analysis. The RMS value is used as a limitation when performing sizing optimization. Based on the research conducted, there was a slight change in thickness in 4 types of rods which were used as design variables, the largest percentage change occurred in the 3 mm stem, namely 0.0866%. The increase in volume was 0.007633588% from 0,1048 m^3 to 0,104808 m^3.

KEYWORDS: Optimization of Sizing, Human Comfort, ISO-2631, RMS Acceleration & Bus Frame Structure

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INTRODUCTION

In several decades, urban areas around the world, in both developed and developing countries, are increasingly dominated by vehicles. In developing countries in particular, cities have experienced rapid growth in transportation-related challenges, including pollution, congestion, accidents, reduced public transport, environmental degradation, climate change, energy depletion, visual intrusion, and lack of accessibility for the urban poor. Today, around the world trying to meet urban standards by improving public transportation[1].

At this time, the conceptual design stage, it is very important to assess various design configurations to the optimized design. A slight reduction in overall weight will result in huge operating cost savings. Thus, it is very important to use optimization tools effectively to determine optimal dimensions of Channels and pipes.Therefore, many researchers have carried out structural optimization to get the lightest vehicle weight possible without reducing the vehicle's performance[2]–[6].

For this reason, the author uses previous research as a result of initial analysis to be used as sizing optimization limits on the bus frame structure by considering passenger comfort with the help of Finite Element Analysis (FEA) software, namely ALTAIR 2019. Finite Element Method (FEM) was used in this optimization

process. This method promised many advantages such as cheap in efficient in cost and times since to conduct the analysis, researcher do not need any real model or prototype. Many researchers used FEM as a tool to optimize the structure of vehicles[7]–[10][2]–[5], [11]–[15].

METHOD

In this study, the object used is a bus design approach from the original design obtained from body builder manufacturer. Like a car body. The thing that must be done before doing the analysis is modeling the bus structure. Figure 1 shows the bus body model.

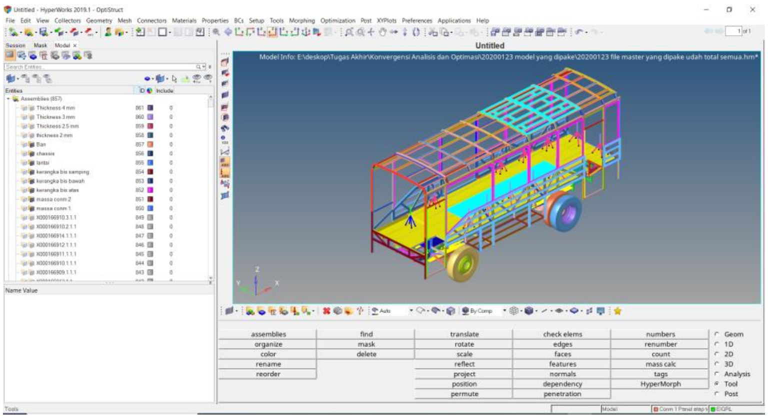


Figure 1: Bus Chassis Modeling.

The non-structural mass inputs contained in the bus are doors, air conditioners, roof panels, doors, roof ceilings, glass, cowl, luggage, floors and the mass of passengers. Set the nodes to position the lumpy mass.In this study, the optimization of the bus structure uses a type of size optimization, where the thickness of the bus frame structure is a parameter in the optimization. There are several steps required to run the optimization, including design variables are numerical inputs that are allowed to change during design optimization. In this research, the thickness of the bus frame will be changed so that it is a design variable. The design variable chosen is a component that uses Stalatube pipes as shown in figure 2.

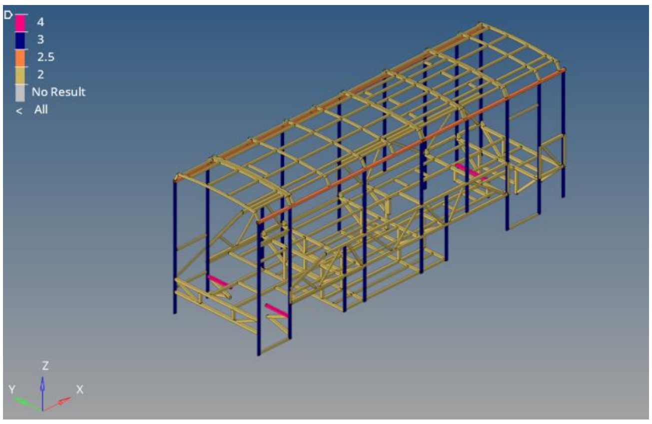


Figure 2: Model for Variable Design.

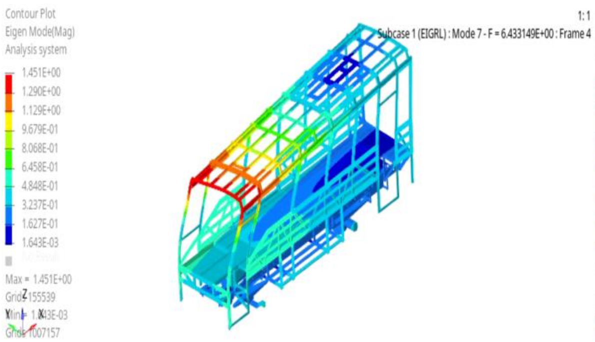
Response optimization helps identify combinations of variable settings that together optimize a single response or series of responses. This is useful when it is necessary to evaluate the impact of multiple variables on a response. In this

study, 19 responses were used. Optimization constraint is a variable that optimizes the objective function with respect to several variables with constraints on these variables. The determination of the value of the constraint is based on an acceleration that is comfortable for humans. This has been studied by Katu et al. [16] based on ISO 2631, it was observed that the critical range that affects humans is in the range 4 to 8 Hz. To analyze the effect of the rate of acceleration of vibrations on humans, it was decided to take the acceleration rate at a frequency of 4 to 8 Hz with a maximum limit of acceleration for human comfort, namely 1600 mm / s2. Therefore, in this study, in order to avoid human discomfort, the researcher uses the acceleration value as a constraint. Several researchers also used dynamic performance as constraints in their optimization of structural vehicles[5], [6], [8], [10][17]–[19].

An optimization problem is one in which several functions are maximized or minimized relative to a given set of alternatives. The function to be minimized or maximized is called the objective function. In this study, the mass of the electric bus structure is an objective function, and its function will be minimized.

RESULTS AND DISCUSSIONS

Modal analysis is a method used to determine the inherent characteristics of a structure such as the personal frequency of the structural system, shape mode, the formulation of a mathematical model for the dynamic behavior of a structure. The natural vibrational mode of a system is affected by the rigidity of the structure, and the mass participating with the structure. Normal modes in this study were carried out in free-free conditions and took the vibration mode from a frequency of 0-20 Hz that was shown in figure 3.



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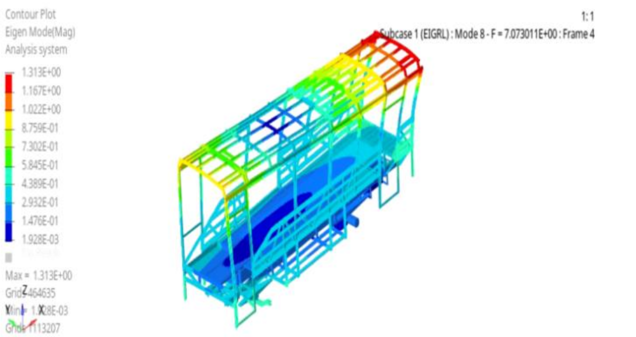

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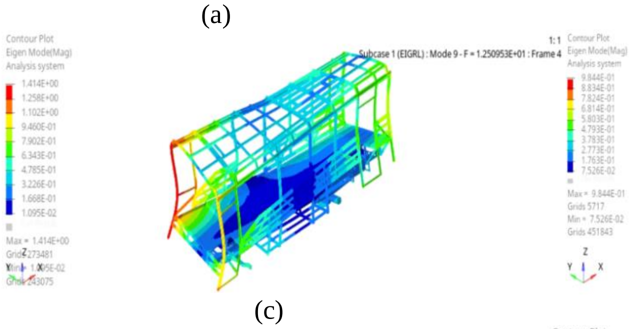

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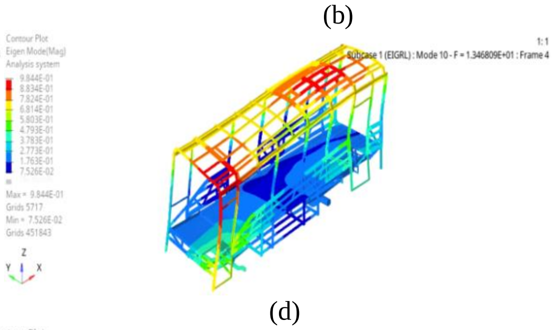

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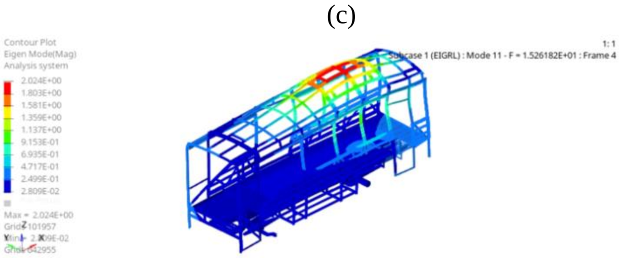

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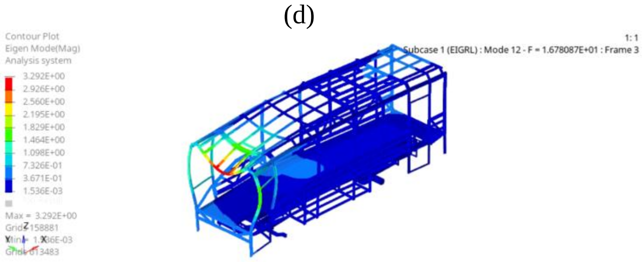

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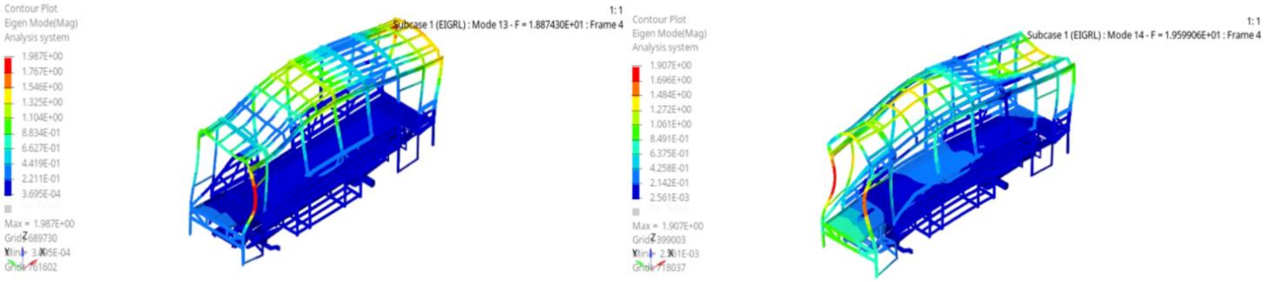


(g) (h)

Figure 3: Value of Shape Mode in (a) Mode 7, (b) Mode 8, (c) Mode 9, (d) Mode 10, (e) Mode 11, (f) Mode 12, (g) Mode 13, and (h) Mode 14.

In the random response analysis process, it can be used the calculated FRF results using Altair Hyperworks software and also the power spectral density profile of the smooth highway roads that are passed so that it can be obtained the dynamic response or PSD output on the front and rear chassis. it was explained that the power spectral density of road unevenness Sg (f) provides input to the vehicle system, namely FRF | H (f) | and produce power spectral density acceleration Sv (f) as output. The input-output relationship of the linear vehicle system is in accordance with equation 1 below:

$$S_v(f) = |H(f)|^2 \, S_g(f) \tag{1}$$

To get the PSD Output, it is necessary to have the FRF Acceleration value as a transfer function that occurs in the driver's seat, middle seat and rear seat. The observed node position lay-out is as follows as displayed in Figure 4.

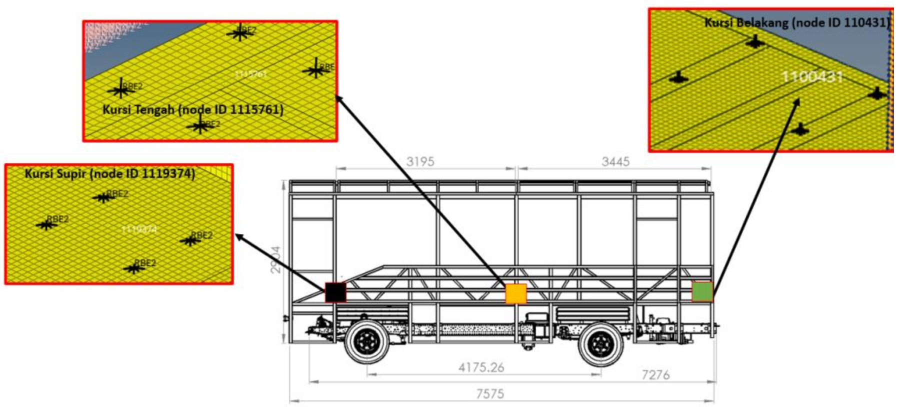
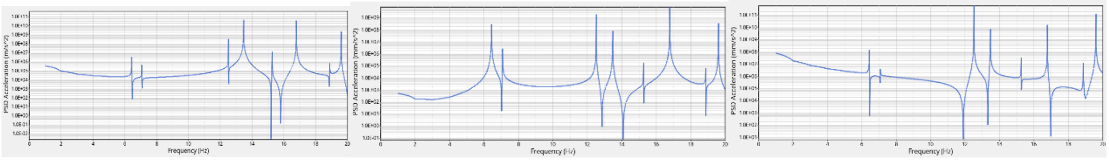


Figure 4: The Position of the Chair Node of Observed Point.

PSD Acceleration due to the effects of road profile excitation and harmonic forces on the front chassis

- PSD output on driver's seat (node 1119374)

After simulating FRF and random response due to harmonic load and the effect of smooth highway road profile, dynamic response or PSD output is obtained as shown in Figure 5 below.

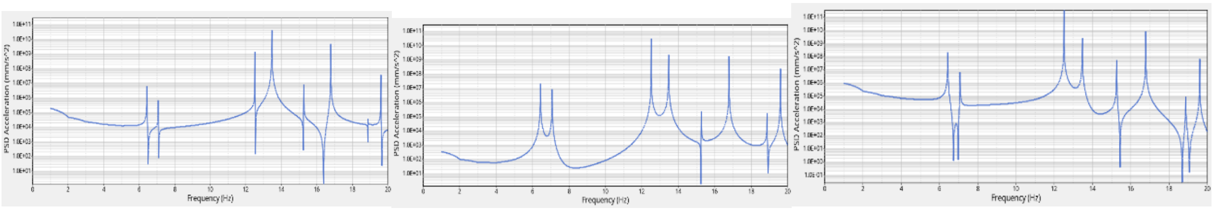


(a) (b) (c)

Figure 5: Dynamic Response Due to Front Chassis Excitation in the Driver's Seat (node 1119374) on (a) X Axis, (b) Y Axis, and (c) Z Axis.

- PSD output on middle seat (node 1115761)

After simulating FRF and random response due to harmonic load and the effect of the smooth highway profile, a dynamic response or PSD output is obtained as shown in Figure 6 below.



(a) (b) (c)

Figure 6: Dynamic Response Due to Front Chassis Excitation in Center seat (Node 1115761) on (a) X-Axis, (b) Y-Axis, and (c) Z-Axis.

Optimization requires the RMS acceleration value obtained from previous research in the form of PSD Acceleration in graphical form. To get the RMS Acceleration value, Altair Hyperworks uses the equation formula (2) as follows:

$$(S(f))_{RMS} = \sqrt{2\int_0^{fn} s(f)\, df} \quad (2)$$

From the results of FRF simulation analysis, random response due to harmonic load, effect of smooth highway profile, PSD output to obtain RMS Acceleration which will be used as response optimization, then optimization of the bus structure with human security acceleration as the constraint so as to obtain the most optimal structure. The optimization method used in this study is the size optimization method. In this optimization, the design variable that is determined is the thickness of the components using the Stalatube pipe. RMS acceleration and volume of the bus structure are used as responses. Acceleration for passenger safety was chosen as constraint for the optimization process. The objective function of this optimization is to reduce the overall mass of the bus structure. The RMS acceleration used the x (lateral), y (lateral), and z (vertical) directions.

This optimization is to obtain a stalatube pipe thickness that is not in accordance with market standards, namely by means of a variable design using a range from 1.5 mm to 4 mm. This optimization is done to get the best thickness. Optimization has been converged. The result is a change in RMS acceleration so that the thickness of the stalatube pipe and the volume of the structure also changes. This design is optimal even though it cannot be manufactured. Changes in RMS acceleration can be seen in table 1. Then the description of the optimization results and the convergence graph of the objective function minimizing volume can be seen in figure 7 and figure 8.

Table 1: Change in RMS Acceleration after Optimization

	RMS Acceleration	Initial Data	Optimization Results	Percentage (%)
1	driver position due to front tire vibration in the X direction	370,18	344,054	-7,057647631
2	driver position due to front tire vibration in the Y direction	1978,54	1657,937	-16,20401913
3	driver position due to front tire vibration in the Z direction	3183,699	3078,727	-3,297170995
4	middle position due to front tire vibration in the X direction	368,737	324,5235	-11,99052441
5	middle position due to front tire vibration in the y direction	621,617	515,5537	-17,06248381
6	middle position due to front tire vibration in the Z direction	1724,194	1451,171	-15,83481905
7	rear position due to front tire vibration in the X direction	280,976	258,3689	-8,045918513
8	rear position due to front tire vibration in the y direction	519,589	421,7311	-18,8337128
9	rear position due to front tire vibration in the Z direction	1071,428	968,3488	-9,620730464
10	driver position due to rear tire vibration in the X direction	25,392	23,6525	-6,850582861
11	driver position due to rear tire vibration in the Y direction	44,539	35,82362	-19,56797414
12	driver position due to rear tire vibration in the Z direction	100,349	97,93837	-2,402246161
13	middle position due to rear tire vibration in X direction	21,575	19,31371	-10,48106605
14	middle position due to rear tire vibration in the y direction	32,589	24,55537	-24,65135475
15	middle position due to rear tire vibration in the Z direction	312,194	311,0201	-0,376016195
16	rear position due to rear tire vibration in the X direction	20,334	18,55677	-8,740188846
17	rear position due to rear tire vibration in the y direction	46,11	35,47164	-23,07169811
18	rear position due to rear tire vibration in the Z direction	413,273	412,789	-0,117113869

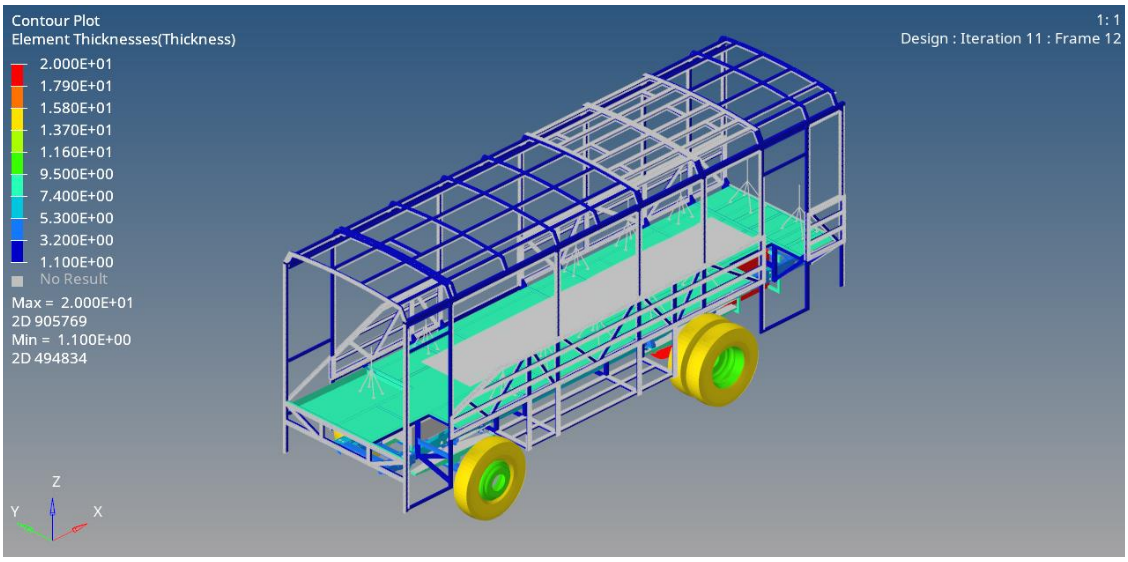


Figure 7: Design Optimization Results.

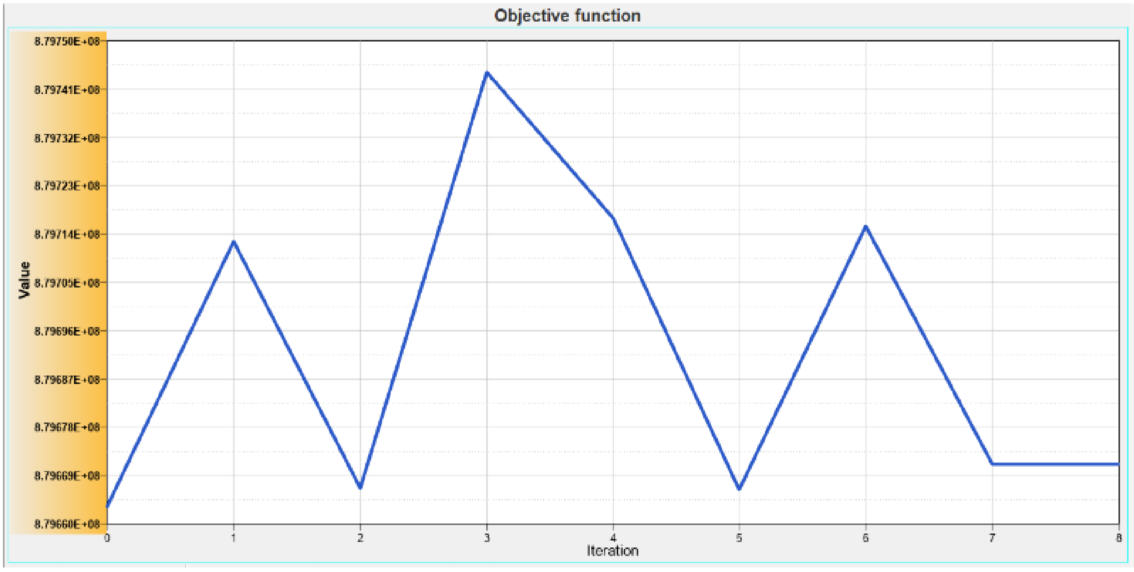


Figure 8: Convergence Graph Objective Function Minimizing Volume.

Changes in pipe thickness and volume of the buss structure after optimization have been shown in Tables 5 and 6 below:

Table 5: Change in Structure Thickness after Optimization

	Bar	Initial Data Thickness (mm)	Optimization Results	Percentage (%)
1	Bar 2 mm	2	1,999536	-0,0232
2	Bar 2,5 mm	2,5	2,501579	0,06316
3	Bar 3 mm	3	3,002598	0,0866
4	Bar 4 mm	4	3,999966	-0,00085

Table 6: Volume Change after Optimization

	Volume	Initial Data (mm3)	Optimization Results	Percentage (%)
1	Overall volume	879663100	879671100	0,000909439
2	Optimized section volume	104800000	104808000	0,007633588

From these results it is convergent but still infeasible design because there are 2 responses that do not meet the constrain rules that have been given, namely 1600 mm / s2 (0.163G).

CONCLUSIONS

The result of frame structure sizing optimization shows that at all positions the RMS acceleration value has decreased. in the lateral direction (x and y) there is still 1 value that exceeds 1600 mm /s^2, namely in the y direction driver position due to the vibration of the front tire of 1657.937 mm /s^2, then in the vertical direction (z) there is 1 value that exceeds 1600 mm / s^2 which is in the driver position due to front tire vibration of 3078.727 mm / s2. On 2 mm rods, the thickness decreases by 0.0232% to 1.999536 mm, for 2.5 mm rods, there is an increase in thickness of 0.06316 % to 2.501579 mm, for a 3 mm stem the thickness increased by 0.0866% to 3.002598 mm, on a 4 mm stem the thickness decreased by 0.00085% to 3.999966 mm. From the results of sizing optimization with a variable design of stem thickness, the volume increase is 0.007633588% from 1.04800000 m^3 to 1.04808000m^3; this indicates that the thickness of the stem before optimization is close to the optimal stem thickness.

ACKNOWLEDGEMENTS

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